FEASIBILITY OF SOLAR-FIRED, COMPRESSOR-ASSISTED ABSORPTION CHILLERS

FEASIBILITY ANALYSIS AND FINAL EISG REPORT

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California Energy Commission
Energy Innovations Small Grant Program

Prepared By:
Bergquam Energy Systems

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ENERGY INNOVATIONS SMALL GRANT (EISG) PROGRAM

FEASIBILITY ANALYSIS REPORT (FAR)

FEASIBILITY OF SOLAR FIRED, COMPRESSOR ASSISTED ABSORPTION CHILLERS

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PREFACE

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Commission), annually awards up to $62 million of which $2 million/year is allocated to the Energy Innovation Small Grant (EISG) Program for grants. The EISG Program is administered by the San Diego State University Foundation under contract to the California State University, which is under contract to the Commission.

The EISG Program conducts four solicitations a year and awards grants up to $75,000 for promising proof-of-concept energy research.

PIER funding efforts are focused on the following six RD&D program areas:

- Residential and Commercial Building End-Use Energy Efficiency
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Environmentally-Preferred Advanced Generation
- Energy-Related Environmental Research
- Strategic Energy Research

The EISG Program Administrator is required by contract to generate and deliver to the Commission a Feasibility Analysis Report (FAR) on all completed grant projects. The purpose of the FAR is to provide a concise summary and independent assessment of the grant project using the Stages and Gates methodology in order to provide the Commission and the general public with information that would assist in making follow-on funding decisions (as presented in the Independent Assessment section).

The FAR is organized into the following sections:

- Executive Summary
- Stages and Gates Methodology
- Independent Assessment
- Appendices
    - Appendix A: Final Report (under separate cover)
    - Appendix B: Awardee Rebuttal to Independent Assessment (Awardee option)

For more information on the EISG Program or to download a copy of the FAR, please visit the EISG program page on the Commission's Web site at:
http://www.energy.ca.gov/research/innovations

or contact the EISG Program Administrator at (619) 594-1049 or email
eisgp@energy.state.ca.us.

For more information on the overall PIER Program, please visit the Commission's Web site at
http://www.energy.ca.gov/research/index.html.

Feasibility Of Solar Fired, Compressor Assisted Absorption Chillers

EISG Grant # 99-15

Awardee:	Bergquam Energy Systems
Principal Investigator:	James Bergquam
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Grant Funding:	$75,000
Grant Term:	January 2000 – July 2001

Introduction

This project researched the feasibility of adding small vapor compressors to solar-fired absorption chillers to lower the operating temperature of the primary generator, simplify the maintenance, and reduce the cost of solar-powered absorption HVAC systems.

The nominal generator temperature in a single-effect absorption chiller is 190^{o}F, and the coefficient of performance (COP) is between 0.7 and 0.8. Standard double-effect absorption chillers require the high temperature 1st stage generator to operate above 300^{o}F. The nominal double-effect cycle COP is 1.2. Various modifications have been proposed to lower the generator temperature in absorption chillers. Any modifications must reduce the operating temperatures of the generators while maintaining the COP of the units. One proposal is to add a small vapor compressor to the basic cycle.

This project involved both the bench testing of an automotive turbocharger and the computer modeling of single effect (1E) and double effect (2E) compressor-assisted absorption chillers. The bench testing showed a turbocharger driven by steam at 240-250^{o}F was able to provide the volumetric flow rate required by a compressor-assisted chiller. Additional work needs to be completed to develop mechanical compressors and/or thermocompressors that are optimized for this application.

The researcher developed computer models to simulate the effect of vapor compressors at selected locations in single and double effect $LiBr/H_2O$ absorption chillers. Two locations for single-effect chillers and three locations for double-effect chillers were modeled. The best results were obtained for a double-effect chiller with the compressor located between the high temperature and the low temperature generators.

Objectives

The goal of this project was to determine the feasibility of using small vapor compressors to reduce the cost, improve the performance, and simplify the maintenance of solar-powered absorption HVAC systems. The researchers established the following project objectives:

1. Develop computer models to simulate the operation and calculate the performance of solar-fired compressor assisted (SFCA) single-effect and double-effect absorption chillers with a lithium bromide/water ($LiBr/H_2O$) solution as the working fluid.
2. Determine the location of the compressors that will provide the greatest increase in the temperature of the refrigerant in the cycle and thereby lower the required operating temperature of the generator.
3. Bench test a small compressor of the type that could be used in this application.

Outcomes

1. The researcher developed computer models to simulate the operation and calculate the performance of SFCA absorption chillers. The computer models determine all of the state points of the LiBr/H_2O solution in the cycle. The specified parameters are the temperatures of the chilled water and the cooling tower, the cooling capacity of the chiller, the concentration of the strong solution, and the compression ratio and compressor efficiency. The models calculate the properties at all of the state points and the coefficient of performance of the chillers and the flow requirements of the vapor compressors.
2. The project calculated results for two compressor locations for a single-effect chiller and three compressor locations for a double-effect chiller. For each compressor location results from the computer models show:
    a. The compression ratio and inlet volumetric flow rate
    b. The outlet temperature and power requirement of the compressor
    c. The operating temperature of and the heat input to the generator
    d. The chiller COP.
3. The researcher bench tested a turbocharger from a 1990 Mazda 626, 2.2 liter engine. From those tests the researcher determined compression ratio, volumetric flow rate, power input, and isentropic efficiency of the compressor.

Conclusions

1. The model performed adequately, although some cases of instability arose, particularly in "Task 7, Evaluation of the 2E Chiller" with the compressor located between the evaporator and the absorber. While the failure had little effect on this study, it creates doubts concerning the value of the model.
2. The calculations indicated the best location for a compressor in a 1E absorption chiller was between the generator and the condenser, though there was concern about the high volume flow requirements. The indicated best location of the compressor in a 2E absorption chiller was between the high temperature generator and the low temperature generator. In this case the high temperature generator can operate at less than 250 ^{o}F, with pressure just below one atmosphere. The compressor outlet temperature and volume flow rate are reasonable for a small compressor, and the overall COP is good. Any future work should focus on this case.
3. The bench tests of a small compressor emphasize the need for a systems engineering approach to the design of a compressor-assisted chiller. For example, the turbocharger was not the right device to use for this purpose. The researcher made no effort to address issues such as air leakage passed the turbine shaft. Rather, the bench testing simply recreated performance data known to the turbocharger manufacturer. A more useful result may have been obtained if the computer model had incorporated a parameterized model of the compressor. Then the researcher could have obtained ideal compressor parameters, allowing him to purchase a compressor built to those specifications.

Benefits to California

Solar-absorption air conditioning is a renewable, non-polluting, environmentally friendly technology. It has the potential to displace conventional, electrically driven compression air conditioners, which is a major consumer of electricity statewide. The implementation of this technology has the potential to significantly reduce both the consumption of and the peak demand for electrical power. This will benefit all the residents of California.

Recommendations

This project investigated the potential of using small vapor compressors in absorption chillers. The researcher identified the operating conditions of the Mazda turbocharger as similar to those useable in absorption chiller systems.

Additional research is necessary to optimize the design of mechanical compressors for solar cooling technology. Another approach is the use of thermocompressors to perform the required compression process. An R&D partner or a commercializing partner should be committed to this project before further public funds are committed.

The next steps are:

1. Identify and acquire a specific absorption chiller in which to incorporate a vapor compressor
2. Perform a detailed specification of a vapor compressor that has the exact specifications required in this application using a trusted computer model
3. Purchase a compressor optimized for absorption chillers that satisfies the specifications for performance.
4. Assemble and bench test the compressor-assisted absorption chiller.

Stages and Gates Methodology

The California Energy Commission utilizes a stages and gates methodology for assessing a project's level of development and for making project management decisions. For research and development projects to be successful they need to address several key activities in a coordinated fashion as they progress through the various stages of development. The activities of the stages and gates process are typically tailored to fit a specific industry and in the case of PIER the activities were tailored to be appropriate for a publicly funded energy research and development program. In total there are seven types of activities that are tracked across eight stages of development as represented in the matrix below.

Development Stage/Activity Matrix

	Stage 1	Stage 2	Stage 3	Stage 4	Stage 5	Stage 6	Stage 7	Stage 8
Activity 1								
Activity 2								
Activity 3								
Activity 4								
Activity 5								
Activity 6								
Activity 7								

A description the PIER Stages and Gates approach may be found under "Active Award Document Resources" at: http://www.energy.ca.gov/research/innovations and are summarized here.

As the matrix implies, as a project progresses through the stages of development, the work activities associated with each stage needs to be advanced in a coordinated fashion. The EISG program primarily targets projects that seek to complete Stage 3 activities with the highest priority given to establishing technical feasibility. Shaded cells in the matrix above require no activity, assuming prior stage activity has been completed. The development stages and development activities are identified below.

Development Stages:	**Development Activities:**
Stage 1: Idea Generation & Work Statement Development	Activity 1: Marketing / Connection to Market
Stage 2: Technical and Market Analysis	Activity 2: Engineering / Technical
Stage 3: Research & Bench Scale Testing	Activity 3: Legal / Contractual
Stage 4: Technology Development and Field Experiments	Activity 4: Environmental, Safety, and Other Risk Assessments / Quality Plans
Stage 5: Product Development and Field Testing	Activity 5: Strategic Planning / PIER Fit - Critical Path Analysis
Stage 6: Demonstration and Full-Scale Testing	Activity 6: Production Readiness / Commercialization
Stage 7: Market Transformation	Activity 7: Public Benefits / Cost
Stage 8: Commercialization	

Independent Assessment

For the research under evaluation, the Program Administrator assessed the level of development for each activity tracked by the Stages and Gates methodology. This assessment is summarized in the Development Assessment Matrix below. Shaded bars are used to represent the assessed level of development for each activity as related to the development stages. Our assessment is based entirely on the information provided in the course of this project, and the final report. Hence it is only accurate to the extent that all current and past work related to the development activities are reported.

Development Assessment Matrix

Stages / Activity	1 Idea Generation	2 Technical & Market Analysis	3 Research	4 Technology Develop-ment	5 Product Develop-ment	6 Demon-stration	7 Market Transfor-mation	8 Commer-cialization
Marketing								
Engineering / Technical								
Legal/ Contractual								
Risk Assess/ Quality Plans								
Strategic								
Production. Readiness/								
Public Benefits/ Cost								

The Program Administrator's assessment was based on the following supporting details:

Marketing/Connection to the Market

The project delivered no marketing information or plan. However, the researcher has published papers in the field and may be expected to continue to do so.

Engineering/Technical

The project did not prove conclusively the concept is technically feasible. In addition, there is considerable concern it may not be economically feasible once technical feasibility is proved.

Legal/Contractual

The researcher raised no patent issues during this project.

Environmental, Safety, Risk Assessments/ Quality Plans

Quality Plans include Reliability Analysis, Failure Mode Analysis, Manufacturability, Cost and Maintainability Analyses, Hazard Analysis, Coordinated Test Plan, and Product Safety and Environmental**.** The project delivered none of these plans.

Strategic

This product has no known critical dependencies on other projects under development by PIER or elsewhere

Production Readiness/Commercialization

This project did not identify a commercializing partner.

Public Benefits

Public benefits derived from PIER research and development are assessed within the following context:

- Reduced environmental impacts of the California electricity supply or transmission or distribution system.
- Increased public safety of the California electricity system
- Increased reliability of the California electricity system
- Increased affordability of electricity in California

The primary benefit to the ratepayer from this research is to increase the affordability of electricity in California. This is derived from the use of solar power in the afternoon to reduce the peak load on the electric grid, which is driven by compressive air conditioning.

Appendix A: Final Report (under separate cover)

Appendix B: Awardee Rebuttal to Independent Assessment (none submitted)

Appendix A to FAR 99-15

ENERGY INNOVATIONS SMALL GRANT (EISG) PROGRAM

EISG FINAL REPORT

FEASIBILITY OF SOLAR FIRED, COMPRESSOR ASSISTED ABSORPTION CHILLERS

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The authors acknowledge the contributions of George Kubik to the completion of this project. George is a graduate student in mechanical engineering at California State University, Sacramento. He worked on the Compressor Evaluation Task and performed the literature and equipment searches and conducted the compressor bench tests.

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Abstract

The purpose of this project was to research the feasibility of using small, vapor compressors in conjunction with solar fired absorption chillers. The goals are to lower the temperature of the primary generator, simplify the maintenance and reduce the cost of solar powered absorption HVAC systems.

The nominal generator temperature in a single effect absorption chiller is 190^{o}F and the coefficient of performance is between 0.7 and 0.8. Standard double effect, absorption chillers require the high temperature (1st stage) generator to operate above 300^{o}F. The nominal double effect cycle COP is 1.2. Various modifications have been proposed to lower the generator temperature in absorption chillers. The modifications need to reduce the operating temperatures of the generators while maintaining the COP of the units. One such modification is to add a small vapor compressor to the basic cycle.

This project involved both the bench testing of an automotive turbocharger and the computer modeling of single effect and double effect, compressor assisted absorption chillers. The bench testing showed that a turbocharger driven by 240-250^{o}F steam was able to provide the volumetric flow rate required by a compressor assisted chiller. Additional work needs to be completed to develop mechanical compressors and/or thermocompressors that are optimized for this application.

Computer models were developed that simulate the effect of vapor compressors at selected locations in single and double effect LiBr/H_2O absorption chillers. Two locations were modeled for single effect chillers and three locations for double effect chillers. The best results were obtained for a double effect chiller with the compressor located between the high temperature and the low temperature generators.

Key Words:

Solar, absorption, air conditioning, compressor, chiller, turbocharger, HVAC, generator

Executive Summary

Introduction

Solar air conditioning using water fired absorption chillers is a technology that has the potential to significantly reduce the peak summertime demand for the electricity that powers conventional compression air conditioning equipment. To date, solar cooling has had a minimal impact in California, and throughout the U.S. The main reasons are the:

- High initial cost of water fired absorption chillers and the collectors that provide the solar energy to drive the chillers.
- Relatively low utility rates that have existed in most of the U.S. for many years.

The purpose of this project is to determine and evaluate the feasibility of using small vapor compressors to reduce the cost, improve the performance and simplify the maintenance of solar powered absorption HVAC systems.

In a solar fired, compressor assisted (SFCA) absorption chiller, the thermal energy to generate the refrigerant (H_2O vapor) is provided by an array of solar collectors. A small vapor compressor is added to the basic cycle to compress, and boost the temperature of, the refrigerant at selected points within the cycle. This project researched the feasibility and effectiveness of solar fired, compressor assisted absorption chillers.

Objectives

The objectives of this project are to:

- Develop computer models to simulate the operation and calculate the performance of SFCA single effect and double effect absorption chillers with a Lithium Bromide/water ($LiBr/H_2O$) solution as the working fluid.
- Determine the location of the compressors that will provide the greatest impact on boosting the temperature of the refrigerant in the cycle and, thereby, lower the required operating temperature of the generator.
- Bench-test a small compressor of the type that could be used in this application.

Outcomes

- Computer models that simulate the operation and calculate the performance of SFCA absorption chillers have been developed. Results are presented in this report for two compressor locations for a single effect chiller and three compressor locations for a double effect chiller.
- A turbocharger from a 1990 Mazda 626 with a 2.2 liter engine has been bench tested in our lab. Performance results for the turbocharger are presented in the report.

Conclusions

- The computer models determine all of the state points of the LiBr/H_2O solution in the cycle. The specified parameters are the temperatures of the chilled water and the cooling tower, the cooling capacity of the chiller, the concentration of the strong solution and the compression ratio and compressor efficiency. The models calculate the properties at all of the state points and the coefficient of performance of the chillers and the flow requirements of the vapor compressors.
- A three-phase turbocharger testing program was successfully completed and a lot of good information and operating data were obtained.
- The project has established the potential benefits of using small vapor compressors in absorption chillers. For each compressor location calculated results from the computer models show:
  1. The compression ratio and inlet volumetric flow rate.
  2. The outlet temperature and power requirement of the compressor.
  3. The operating temperature of, and the heat input to, the generator.
  4. The coefficient of performance of the chiller.
- The results of the Bench Testing show the measured compression ratio, volumetric flow rate, power input and the isentropic efficiency of the compressor.

Recommendations

This project has demonstrated the potential of using small vapor compressors in absorption chillers. The operating conditions of the Mazda turbocharger are pretty close to, but not quite the same as, those required by absorption chillers. The reason is that the compressor blades in turbochargers are backward-inclined and require very high speeds to produce the needed pressure ratios and flow rates. Additional research is necessary to optimize the design of mechanical compressors for solar cooling technology. Another approach that needs to be evaluated is the use of thermocompressors to perform the required compression process.

The next steps are to:

- Perform a detailed design of a vapor compressor that has the exact specifications required in this application.
- Specify the size and shape of the compressor blades and the configuration of the compressor.
- Purchase and/or build a compressor that is optimized for absorption chillers.
- Perform a finite element analysis of the flow in a thermocompressor.
- Purchase and/or build and bench test a thermocompressor that is optimized for absorption chillers.

Benefits to California

This project contributed to the Public Interest Energy Research (PIER) program objective of reducing consumption of and demand for electricity in California.

Solar absorption air conditioning is a renewable, non-polluting, environmentally friendly technology. It has the potential to displace conventional, electrically driven compression air conditioners. The implementation of this technology will significantly reduce both the consumption of, and the peak demand for, electrical power. This will benefit all the residents of California.

Introduction

The purpose of this project was to determine and evaluate the feasibility of using small vapor compressors to improve the performance, simplify the maintenance and reduce the cost of solar powered absorption HVAC systems.

The main components of an absorption chiller are the generator(s), condenser, evaporator, absorber and heat exchanger(s) [1]. These air conditioners are referred to as single effect (1E) if they have only one generator. In this case, the nominal operating temperature is 190^{o}F and the maximum coefficient of performance is between 0.7 and 0.8. An absorption chiller with two generators is called double effect (2E). The high temperature generator operates at approximately 300^{o}F and the nominal COP is 1.2.

One important factor in achieving the goals of improved performance, simplified maintenance and lower cost of solar HVAC systems is to lower the operating temperature of the primary generator in the absorption chillers. Various modifications have been proposed to accomplish these goals [2,3]. One such modification is to add a vapor compressor to the basic cycle.

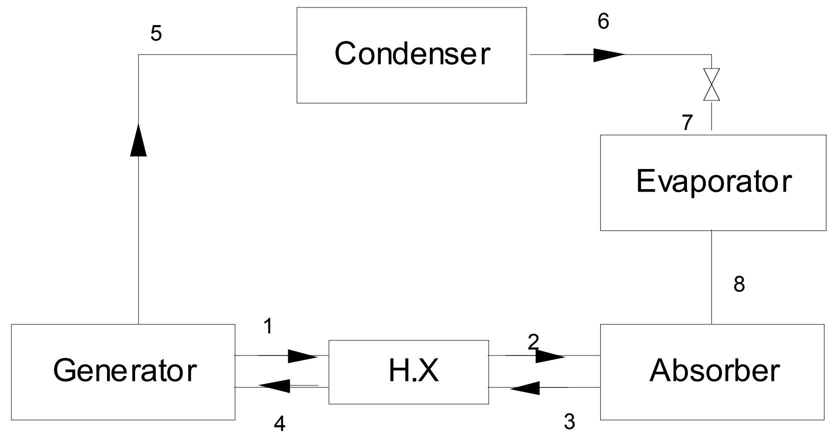
Objectives

The components of a single effect absorption chiller are illustrated in Fig. 1. Also shown are the nominal state points and a PTX diagram for the cycle. As mentioned above, the nominal operating temperature of the generator is 190^{o}F. The Contractor (Bergquam Energy Systems – BES) has been operating commercial, 1E, solar absorption HVAC systems continuously for the past 17 years. It has been demonstrated [4] that both flat plate and parabolic trough collectors can be used to drive 1E chillers. However, the cost of the collector array, which is approximately 40% of the cost of the HVAC system, makes the payback periods of these systems between 7 and 8 years.

One of the objectives of this project was to determine and evaluate the feasibility of SFCA single effect absorption chillers. The most important consideration is where to locate the compressor. Two compressor-assisted configurations have been evaluated. As shown in Fig. 2 and Fig. 3, the two options are:

- Location 1 (Fig. 2) with the compressor between the generator and the condenser.
- Location 2 (Fig. 3) with the compressor between the evaporator and the absorber.

Computer models have been developed to perform a thermodynamic analysis of these two options. The working fluid is a Lithium Bromide/Water (LiBr/H_2O) solution with water as the refrigerant and LiBr as the absorbent. Actual working fluid state points have been used and mass flow and energy balances have been performed on the individual components and on the entire cycle. The objective is to determine the feasibility of these types of single effect absorption chillers.



State point	Temp(F)	press(mmHg)	LiBr (%)
1	191	57.06	0.610
2	126	8.54	0.610
3	105	8.54	0.558
4	160	57.06	0.558
5	191	57.06	0
6	105	57.06	0
7	48	8.54	0
8	48	8.54	0

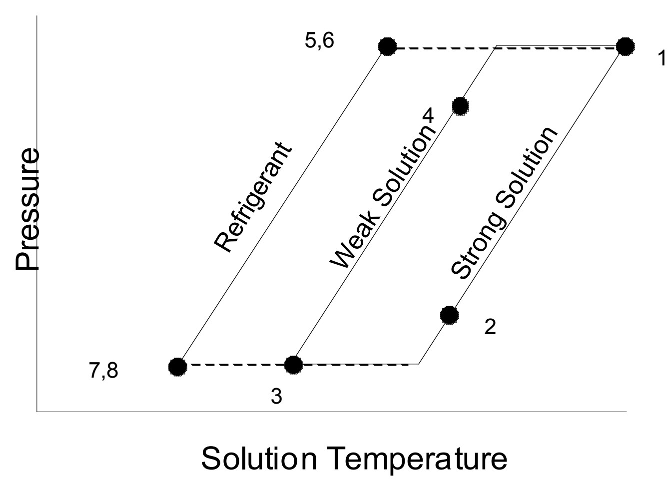


Fig. 1 - Single Effect Absorption Chiller

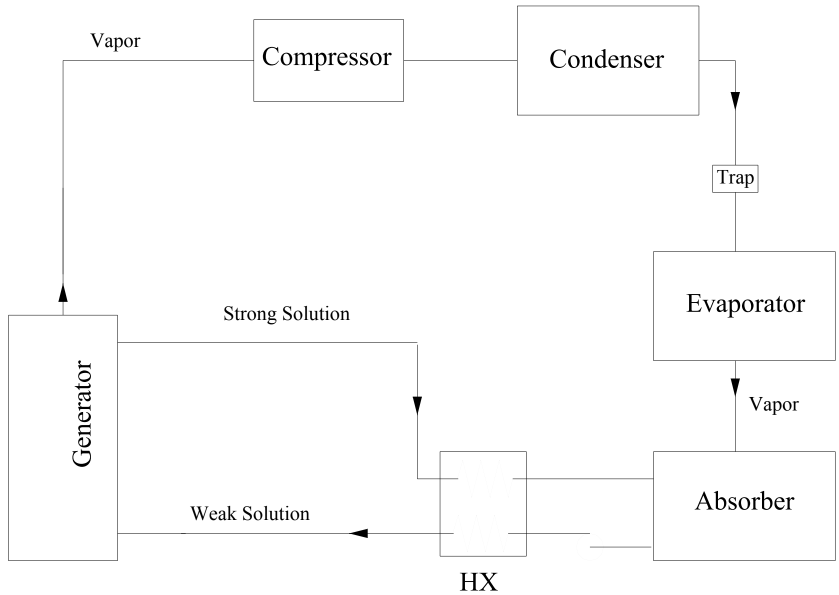


Fig. 2 - 1E with Compressor Location 1

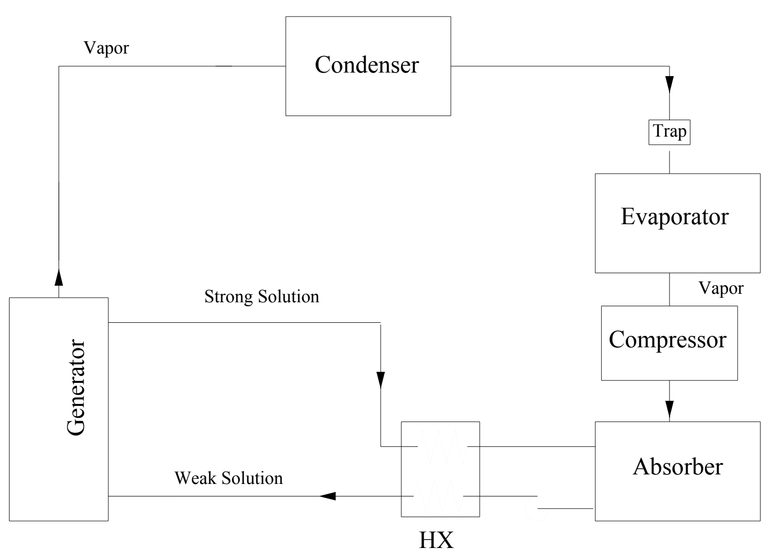


Fig. 3 -1E with Compressor Location 2

Nominal State Points for Double Effect Chiller				
Point	State	Temp (^oF)	Press (mmHg)	Conc. (%wt)
1	Intermediate Soln.	310	686.9	61
2	Intermediate Soln.	199	686.9	61
3	Intermediate Soln.	189	52.6	61
4	Strong Soln.	200	52.6	63.8
5	Strong Soln.	124	52.6	63.8
6	Strong Soln.	124	5.8	63.8
7	Weak Soln.	101	5.8	57.6
8	Weak Soln.	165	-	57.6
9	Weak Soln.	263	686.9	57.6
10	Superheated Vapor	310	686.9	-
11	Saturated Liquid	207	686.9	-
12	Liquid & Vapor	102	52.6	-
13	Saturated Liquid	102	52.6	-
14	Liquid & Vapor	38	5.8	-
15	Saturated Vapor	38	5.8	-
16	Superheated Vapor	200	52.6	-

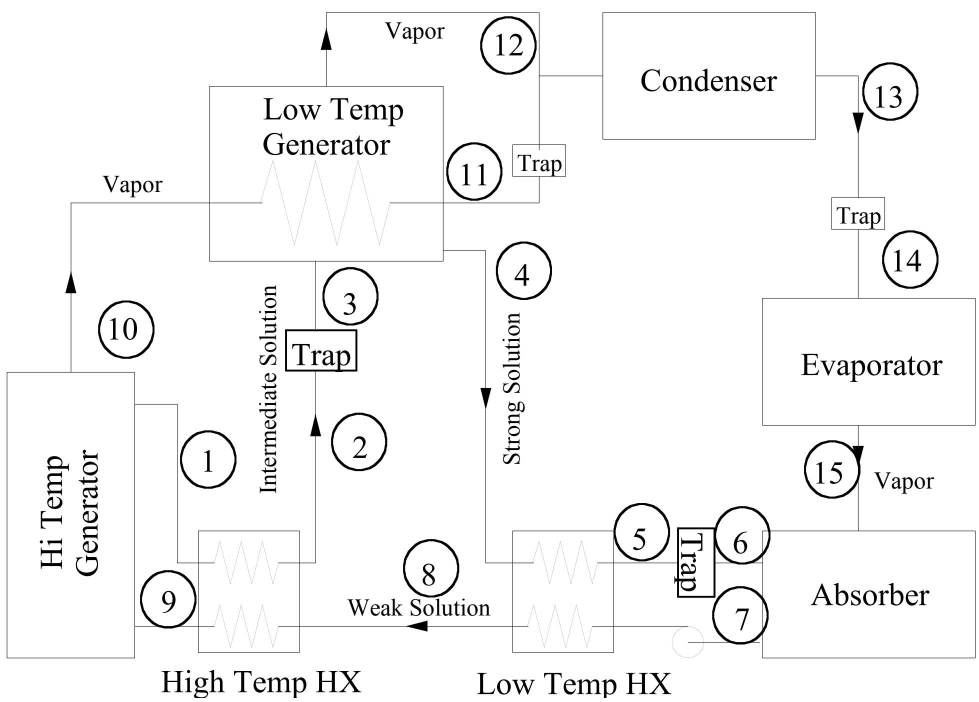


Fig. 4- Double Effect Absorption Chiller

The components of a double effect absorption chiller are illustrated in Fig. 4 together with a list of the nominal state points. The difference between 1E and 2E is that the 2E chillers have a second (low temperature) generator [5]. The work on 2E absorption chillers is based on a 20 ton McQuay/Sanyo double effect chiller that the authors have operated for five years. This unit, and larger chillers manufactured by Trane, both are designed with the generators in series. In the low temperature generator, energy from the hot refrigerant vapor (H_2O) out of the high temperature generator is used to generate additional refrigerant vapor. As a result, the chiller produces more refrigerant, and the evaporator more chilled water, for a given heat input to the high temperature generator. This results in a higher cycle COP. The nominal value is 1.2. However, it also requires a higher temperature, about 300^{o}F, in the high temperature generator.

It has been demonstrated that a 20 ton 2E chiller that was modified from a natural gas driven generator to a water fired, high temperature generator can be operated with heat from an array of vacuum tube solar collectors [6].

Because of the high operating temperatures of 2E chillers, the collector array, the solar storage tank, the pumps, piping and all of the related equipment have to be designed according to pressure vessel codes. This increases the cost of all of these components by as much as a factor of 3. By lowering the operating temperature to under 250^{o}F, the entire system would be classified as a low temperature water (LTW) device.

Another objective of this project was to determine the feasibility of designing and operating a 2E SFCA absorption chiller. The goals are to have an operating temperature under 250^{o}F and a chiller COP between 1.1 and 1.2. This will improve the cost effectiveness of the technology and, with present utility rates, result in a payback period of under 5 years.

As shown in Figs. 5, 6 and 7, three different locations have been evaluated for installation of the vapor compressor. These are:

- Location 1 (Fig. 5) with the compressor between the high temperature generator and the low temperature generator.
- Location 2 (Fig. 6) with the compressor between the low temperature generator and the condenser.
- Location 3 (Fig. 7) with the compressor between the evaporator and the absorber.

As mentioned above, the working fluid in both 1E and 2E absorption chillers is a $LiBr/H_2O$ solution. Especially for the 2E type, the pressures, temperatures and solution concentrations are very important in determining the feasibility of operating the chillers efficiently [7]. Computer models have been developed to perform a thermodynamic and heat transfer analysis of the three options. Actual working fluid state points have been used to perform mass flow and energy balance calculations on the individual components and on the entire cycle. For all of the models, the thermodynamic property data, for both the refrigerant and the $LiBr/H_2O$ solution, are obtained from correlations presented in [7].

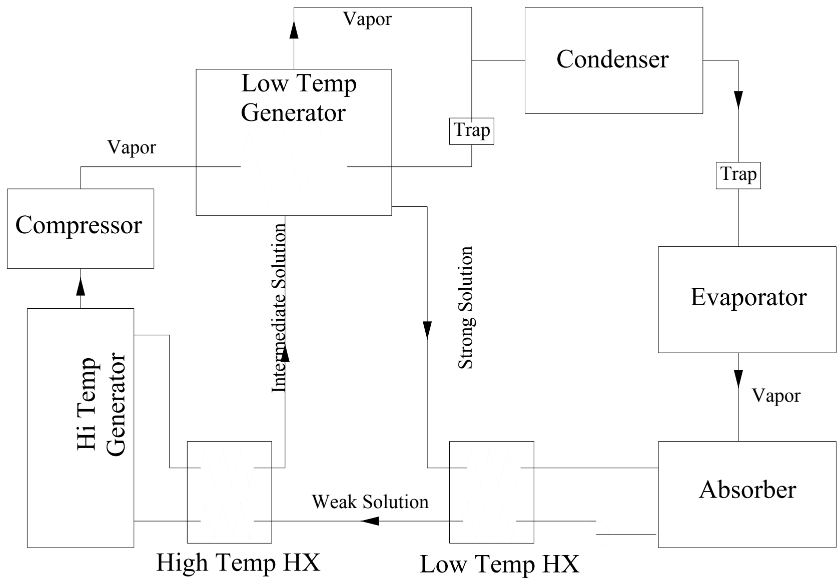


Fig. 5 - 2E with Compressor Location 1

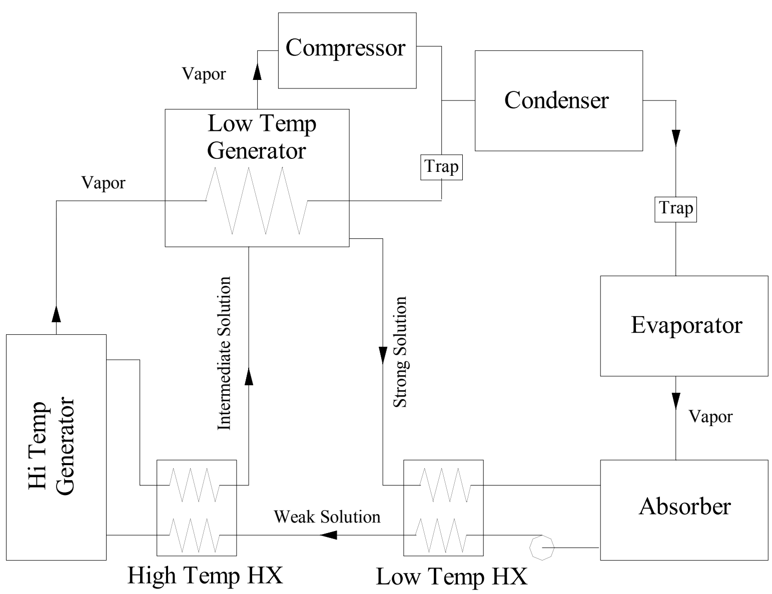


Fig. 6 - 2E with Compressor Location 2

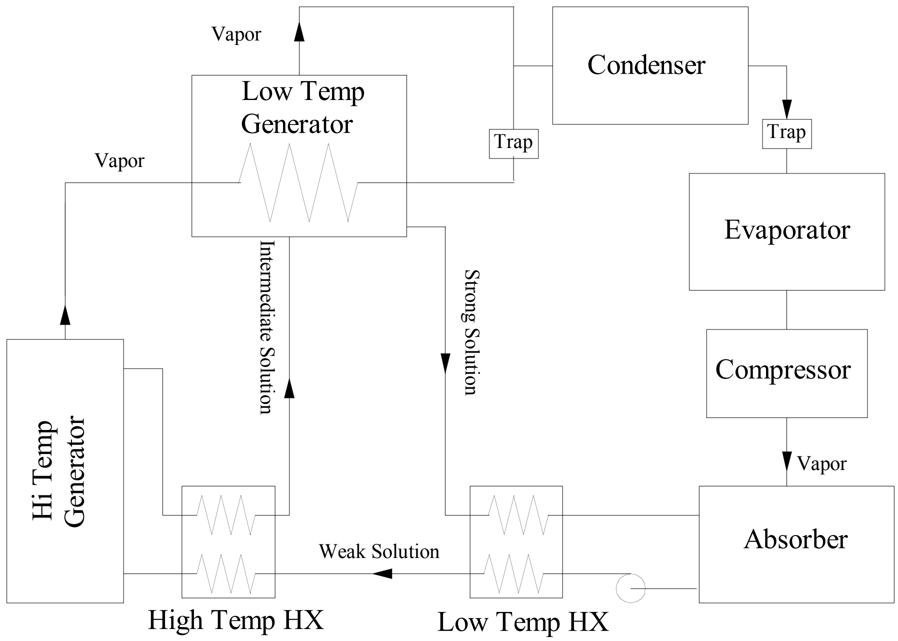


Fig. 7 - 2E with Compressor Location 3

Attached to this report as Appendix I and Appendix II are mass balance, salt (LiBr) balance and energy balance equations for the 1E and 2E absorption chiller models. These equations have been programmed in the computer models.

The appendices start with a schematic showing the components of the chiller and a table describing the state points. The energy and mass balance equations for all of the components are then presented. The equations are solved using the procedures described [8]. After all of the energy conservation equations have been solved, the coefficient of performance (COP) is calculated. The COP is defined as:

$$\mathrm{COP} = \frac{\text{Desired Output}}{\text{Required Input}}$$

$$= \frac{\dot{Q}_{Evap}}{\dot{Q}_{Gen} + \dot{W}_{Comp}}$$

The desired output is always Q_{Evap} and the required input is Q_{Gen} plus the compressor power W_{comp}.

The compressor provides part of the energy input to the chiller. The compressor increases the temperature (and enthalpy) of the refrigerant and even though it is a different energy source, it needs to be included in the COP. It is important to note that the compressor work is small compared to the heat input. However if this was not the case, the COP would be low and the compressor assisted chiller would be impractical.

The third main objective of this project is to bench test a compressor of the type that could be used in a SFCA chiller. The compressors make it possible to lower the operating temperature of the high temperature (1st stage) generators, in both 1E and 2E chillers, without significantly lowering the COP of the cycle.

Our research indicates that the compressors required with absorption chillers are in the operating range of small automotive turbochargers. One of the differences is that standard turbochargers are designed to compress air. The working fluid to be compressed in an absorption cycle is superheated steam at less than atmospheric pressure.

For the bench testing part of the project, we acquired an IHI model FEH5A turbocharger. The unit is from a 1990 Mazda 626 with a 2.2 liter engine. The turbocharger was installed and tested in our shop. A Picture of the turbocharger, as it was installed, is shown in Fig. 8.

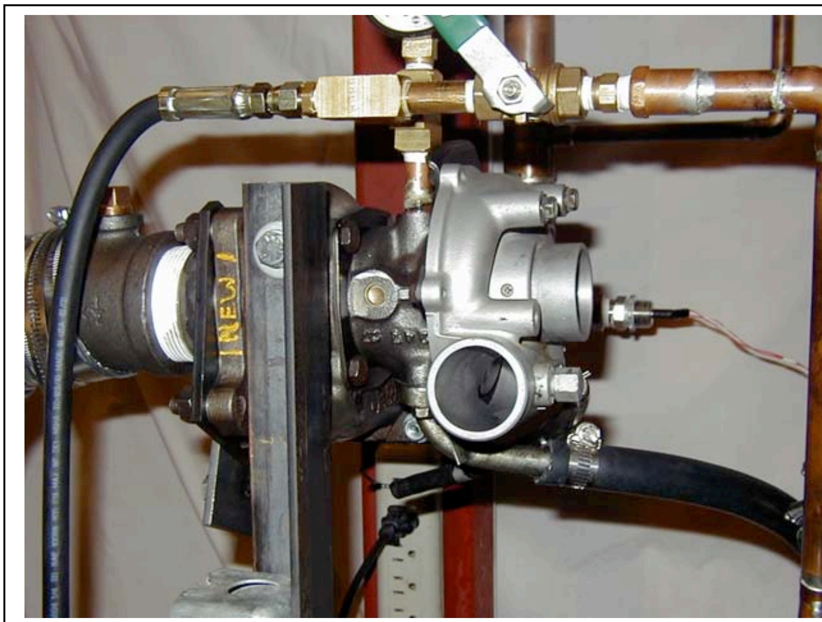


Figure 8 - Turbocharger

Project Approach

In order to accomplish the Project Objective, the project was organized into the following tasks:

Task 1 – Compressor Evaluation

Bergquam Energy Systems performed Task 1under the direction of Dr. James Bergquam, Principal Investigator, and Mr. Joseph Brezner, Lead Engineer. Mr. George Kubik did the literature and equipment searches and conducted the tests. The primary goal of Task 1 was to acquire a candidate compressor and build a device to test the operation of the compressor. The bench testing was performed on a turbocharger from a 1990 Mazda.

The turbocharger was used because it provides the required pressure ratio and volumetric flow rate. The authors recognize that there is a potential problem involving air leakage with a turbocharger. Because of this, alternative methods of driving the compressor and of performing the compression process need to be evaluated.

The initial testing was performed, by driving the turbine side of the turbocharger with air from a 10 hp compressor. The second phase of the testing program involved operating the turbine with air from a jackhammer compressor. For the third phase, the turbine was driven by 250^{o}F steam from an existing 1,000 gal., insulated storage tank.

Task 2 – Develop Computer Models

Task 2 was performed by Mr. Brezner and Dr. Bergquam. Conservation of mass and conservation of energy equations were used to develop computer models for single effect and double effect absorption chillers. The models use operating pressures and temperatures to determine the properties of the working fluids and all of the state points in the cycles. The models are used for conventional and compressor assisted absorption cycles. Two compressor locations are considered for single effect cycles and three compressor locations for double effect cycles.

Task 3 – 1E Chiller – Compressor Location 1

Task 3 was performed by Mr. Brezner and Dr. Bergquam. For this case the compressor is located between the generator and the condenser. The computer models gave good results for the generator temperature, the compressor outlet temperature and the COP of the cycle. However, the compressor requires a large inlet volume flow rate. This may not be attainable by a small vapor compressor. This case would warrant further investigation for situations where a source of inexpensive, low-grade heat is available. One possible application is to use low temperature thermal energy from a PV array.

Task 4 – 1E Chiller – Compressor Location 2

Task 4 was performed by Mr. Brezner and Dr. Bergquam. Location 2 has the compressor between the evaporator and the absorber. The pressure, and therefore the density, of the vapor leaving the evaporator and entering the compressor are very low. This results in an even larger flow rate at the inlet to the compressor. The model gives good results for all of the chiller parameters but the high volumetric flow rate makes this location impractical.

Task 5 – 2E Chiller – Compressor Location 1

Task 4 was performed by Mr. Brezner and Dr. Bergquam. Location 1 for the double effect chiller is between the high temperature generator and the low temperature generator. Of the chiller modifications studied, this one gave the best results. The analysis indicates that the high temperature generator can operate at less than 250^{o}F and at a pressure just below 1 atmosphere. The outlet temperature of the compressor is reasonable, the volume flow rate is attainable by a small compressor and the COP of the unit is good. Future work needs to focus on this case. Both steam driven and motor driven, mechanical compressors and steam driven thermocompressors need to be investigated.

Task 6 – 2E Chiller – Compressor Location 2

Task 6 was performed by Mr. Brezner and Dr. Bergquam. For the 2E chiller, compressor Location 2 is between the low temperature generator and the condenser. This is the second best location for the vapor compressor. The flow rates are similar to those for the single effect, Location 1 case. However, the COP is much higher but the generator temperature is also higher. Additional research needs to be done for this case.

Task 7 – 2E Chiller – Compressor Location 3

Task 7 was performed by Mr. Brezner and Dr. Bergquam. Location 3 is between the evaporator and the absorber. The conclusions for this case are similar to those for the 1E Chiller – Compressor Location 2. There is not enough of a reduction in the generator temperature, and the volumetric flow is too large. This case is impractical.

Project Outcomes

Task 1 – Compressor Evaluation

For this part of the project, an IHI model FEH5A turbocharger from a 1990 Mazda 626 with a 2.2 liter engine was bench tested in our shop. Pictures of the turbocharger and the test device are shown in Figs. 8, 9 and 10. The turbine and compressor wheels are both two inches in diameter. The compressor wheel is made of an aluminum alloy and has backward-inclined blades.

The initial testing of the turbocharger was done using a 10hp air compressor to drive the turbine. These tests were unsuccessful due to insufficient volumetric flow rate from the air compressor. The maximum flow rate produced by the air compressor was not enough to spin the turbocharger at a rate that produced the necessary pressure ratio and flow rate from the compressor side of the turbocharger.

In order to spin the turbocharger faster, a 50hp jackhammer compressor was rented and connected to the turbocharger for a second phase of bench testing. Instrumentation, as shown in Fig. 9, was installed on the test device. The thermocouples were connected to a data acquisition board, which was connected to a PC. Data were recorded every minute. A pitot tube was also installed in the outlet line from the turbocharger compressor to measure velocity. A manometer was connected across the pitot tube. Manometer readings, in inches of H_2O, were converted to velocities and volume flow rates.

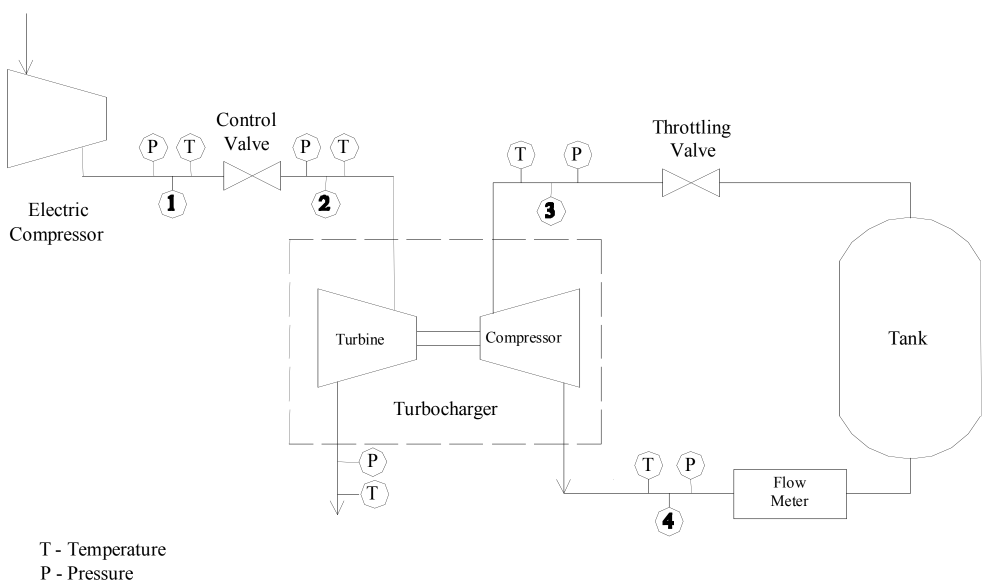


Figure 9 - Turbo Test Setup (Compressed Air)

Results from the second phase of the Bench Testing are presented in Table 1. Values of compressor efficiency, compression ratio, outlet flow and compressor power consumption are listed for each manometer reading.

Table 1 – Phase Two Bench Testing

MANOMETER HEIGHT in H_2O	COMP. EFFICIENCY %	PRESSURE RATIO P2/P1	OUTLET FLOW RATE CFM	COMPRESSOR POWER HP
1	80.9	1.20	52	0.75
2	95.6	1.21	72	0.96
3	92.6	1.17	89	1.01
4	93.8	1.16	102	1.05
6	83.7	1.12	125	1.16
8	83.5	1.11	144	1.22
10	64.8	1.10	162	1.26

The power, required by the turbocharger compressor, is the product of the mass flow rate and the sum of the enthalpy change and the kinetic energy change. The compressor efficiency is the ratio of the calculated isentropic compressor power and the actual compressor power.

The second phase of the bench testing demonstrated that the turbocharger is capable of producing the required volumetric flow rate. However, having to drive the turbocharger with a jackhammer compressor is not practical or economical.

It also became apparent that the best method to drive the turbocharger is to use high temperature steam. In solar cooling applications, there is a potential source of 240°F-250°F steam from the storage tank. The third, and final phase of the bench testing was to drive the turbocharger with steam.

The third phase of the Bench Testing was done using a 1,000 gallon, insulated storage tank filled with 750 gallons of water and 250 gallons of steam at 250°F and 30 psia. One and a half inch copper tubing was plumbed from the storage tank to the turbocharger. Compressed air and steam were throttled and mixed before entering the turbine. The setup of pressure gages, thermocouples and pitot tube and manometer was the same as in the previous tests. Fig. 10 is a schematic of the test setup. The tank shown is a 6 cubic foot steel tank connected to the compressor side of the heat exchanger. The water in the storage tank was heated using an array of evacuated tube solar collectors and a back-up boiler.

Table 2 - Phase Three Bench Testing

MANOMETER HEIGHT in H_2O	COMP. EFFICIENCY %	PRESSURE RATIO P2/P1	INLET FLOW RATE CFM	COMPRESSOR POWER HP
1	80.4	1.30	63	1.12
4	77.8	1.23	118	1.87
12	58.2	1.10	184	2.06
14	54.9	1.11	203	2.60

Results from the Phase Three Bench Testing are shown in Table 2. During the test, the storage tank was heated so the temperature of the steam was constant. The valve between the steel tank and the compressor inlet was adjusted to change the flow rate through the compressor. During the testing, data were recorder in 1-minute time intervals. The data in Table 2 are average values taken at four different manometer readings and flow rates. When driven by steam, the turbocharger reached a steady state condition and was able to provide good compressor flows and acceptable pressure ratios.

The Phase Three testing was successful and provided good operational data. The next steps are to optimize the design of mechanical compressors for the pressure ratios and flow rates required by absorption chillers and to conduct research into the use of thermocompressors for this application.

Figure 10 - Turbo Test Setup (Steam and Compressed Air)

Task 2 – Computer Models

The computer models use the conservation of mass and conservation of energy equations to calculate the temperatures, pressures, mass flow rates, heat transfer rates and the COP of the chiller for each system. The chiller uses LiBr as the absorbent and H_2O as the refrigerant. The absorption chiller model uses the pressure/temperature relationships of the absorbent/refrigerant pair to determine the LiBr concentration and the state points within the cycle. Mass balance and energy conservation equations are written for each component of the chiller. The model assumes that the generator is water fired and that the absorber and condenser are cooled by a cooling tower. The recuperators are plate type heat exchangers with an assumed effectiveness of 0.8.

Standard operating conditions and a capacity of 20 tons (240,000 Btu/hr) are used for all of the results presented in this section. The standard conditions are:

Cooling Capacity	20 tons
Evaporator Temperature	45^{o}F
Cooling Tower Temperature	85^{o}F
HX Effectiveness	0.8
Compressor Efficiency	1.0
Heat Transfer ΔT (all components)	10^{o}F

In general, the compression ratio of the vapor compressor is varied from 1 to 4. For some of the cases the models become unstable at compression ratios greater than 2.0. In these cases, results are not presented for larger compression ratios. This is a stability limitation on the model. It appears to be the caused by state point values being outside the range of the property table.

Task 3 – 1E Chiller Compressor Location 1

The first calculations were to determine the chiller COP as a function of compression ratio (CR) for an isentropic compressor (compressor efficiency = 1). Table 3 shows the results for compression ratios from 1 to 4. It is possible to lower the generator temperature below the nominal 191^{o}F and still maintain a COP of 0.75 or 0.76. The compressor work is small in comparison to the generator heat input. In order for the chiller to operate properly, the compression ratio must be increased as the generator temperature (and pressure) are lowered. The lower pressure reduces the density of the steam as it enters the compressor. This required higher compressor inlet volumetric flow in order to maintain the required mass flow. The best conditions are for the CR = 2 with T_{gen} = 164^{o}F, COP = 0.76 and the inlet flow = 2,667 cfm. The problem with this case is the large volume flow.

Table 3 - Single Effect Chiller (Compressor Location 1)

C.R	Tgen ^{o}F	T_{comp} ^{o}F	COP	Wcomp Btu/hr	Qgen Btu/hr	Compressor Inlet cfm
1	191	191	0.76	0	314,081	1,389
1.5	175	242	0.76	7,191	307,914	2,034
2	164	281	0.76	12,560	303,733	2,667
2.5	156	313	0.76	16,894	300,594	3,291
3	150	341	0.75	20,554	298,094	3,909
3.5	145	372	0.75	23,737	296,024	4,560
4	140	386	0.75	26,564	294,284	5,129

Task 4 – 1E Chiller Compressor Location 2

The results from the computer models for this case are summarized in Table 4. The largest compression ratio for which the model is stable is 2.5. In this case the compressor is located between the evaporator and the absorber. The low pressure, low-density vapor leaving the evaporator is input to the compressor. Since the evaporator conditions are fixed in the model, the volume flow rate at the compressor inlet is always the same.

Again, look at the case with the CR = 2. The generator temperature is 164°F, the outlet temperature from the compressor is 143°F and the COP is 0.80. These values are all reasonable. However, the cfm at the compressor inlet is 2.7 times greater than Location 1. Because of this, Location 2 is not realistic.

Table 4 - Single Effect Chiller (Compressor Location 2)

C.R	Tgen °F	Tcomp °F	COP	Wcomp Btu/hr	Qgen Btu/hr	Compressor Inlet cfm
1	191	48	0.76	0	314,081	7,251
1.5	175	102	0.78	5,662	299,250	7,251
2	164	143	0.80	10,062	287,787	7,251
2.5	155	177	0.82	13,714	278,741	7,251

Task 5 – 2E chiller Compressor Location 1

Results from the computer models for a 20 ton, 2E chiller with the compressor between the high temperature generator and the low temperature generator (location 1) are presented in Table 5. For the 2E chillers, the generator temperature is calculated in the model as a function of the compression ratio. In addition, the 1st stage generator operates at close to atmospheric pressure. As a result, the density of the vapor is large enough that the compressor inlet volumetric flow is reasonable.

Consider the case for the CR = 3. The model predicts a generator temperature of 245°F, a compressor flow of 203 cfm and a chiller COP of 1.36. The important points are that the COP is high and the compressor cfm is low.

The reason for using a compressor is to lower the generator temperature below 250°F and still maintain the double effect operation. These are good results indicating that a compressor assisted 2E chiller with compressor Location 1 may be a viable approach for a SFCA absorption HVAC system.

Table 5 - Double Effect Chiller (Compressor Location 1)

C.R	Tgen °F	Tcomp °F	COP	Wcomp Btu/hr	Qgen Btu/hr	Compressor Inlet cfm
1	300	300	1.38	0	173931	76
1.5	280	350	1.38	4635	170570	108
2	265	401	1.37	8047	167502	141
2.5	253	435	1.36	10752	165062	174
3	245	465	1.36	13044	163170	203
3.5	236.5	490	1.36	14971	161421	237
4	231	510	1.35	16716	160038	266

Task 6 – 2E Chiller Location 2

Calculations from the computer models for this case are presented in Table 6. The compression ratio was varied from 1 to 4. The generator and compressor temperatures and the COP are all good. However, the compression ratio needs to be greater than 3 in order to lower the generator temperature to below 250°F. In addition, the required compressor cfm is too large to be achieved in practice.

Table 6 - Double Effect Chiller (Compressor Location 2)

C.R	Tgen °F	Tcomp °F	COP	Wcomp Btu/hr	Qgen Btu/hr	Compressor Inlet cfm
1	300	300	1.38	0	173,931	820
1.5	282	247	1.37	3287	171,845	1217.5
2	270	287	1.36	5795	169,871	1661.47
2.5	261	319	1.36	7849	168,363	2002.48
3	252	346	1.36	9600	167,089	2391.12
3.5	246	371	1.35	11137	166,113	2778.1
4	241	393	1.35	12512	165,295	3163.48

Task 7 – 2E Chiller Location 3

The results from the computer model for this case are summarized in Table 7. The largest compression ratio for which the model is stable is 2. The discussion for Task 4 (1E chiller, compressor location 2) also applies to this case. The model indicates that it is not possible to lower the generator temperature below 270°F. As before, the large compressor cfm makes this location impractical.

Table 7 - Double Effect Chiller (Compressor Location 3)

C.R	Tgen ^oF	Tcomp ^oF	COP	Wcomp Btu/hr	Qgen Btu/hr	Compressor Inlet cfm
1	300	300	1.38	0	173931	8010
1.5	282	247	1.37	3287	171845	8010
2	270	287	1.36	5795	169871	8010

Conclusions and Recommendations

This project resulted in the following conclusions:

- Computer models of compressor assisted, single effect and double effect absorption chillers have been developed. The models determine all of the state points of the LiBr/H_2O solution in the cycles.
- The models calculate the properties at all of the state points and the coefficient of performance of the chillers and the flow requirements of the vapor compressors.
- The models indicate that compressor assisted absorption chillers can operate at significantly lower generator temperatures without adversely affecting the COP.
- A three phase testing program, using a Mazda Turbocharger, was successfully completed. A lot of good information and operating data were obtained.
- The project has established the potential benefit of, and the best location for, small vapor compressors in absorption chillers.
- The results of the Bench Testing show the measured compression ratio, volumetric flow rate, power input and the isentropic efficiency of the compressor side of the turbocharger.

The project has demonstrated the potential of using small vapor compressors in absorption chillers. Additional research is necessary to optimize the design of mechanical compressors for solar cooling applications. Another approach that needs to be evaluated is the use of thermocompressors to perform the required compression process.

With the double effect, compressor assisted chiller the COP should be in the range from 1.0 to 1.1. This compares to a maximum COP of 0.7 for a standard single effect chiller. The higher COP reduces the required collector area by 30-40%. This results in a significant reduction in the cost of the 2E system.

The next steps are to:

- Perform a detailed design of a vapor compressor that has the exact specifications for this application.
- Specify the size and shape of the compressor blades and the configuration of the compressor.
- Evaluate and compare compressors that are driven by steam and by electric motors.

- Purchase and/or build a compressor that is optimized for absorption chillers.
- Perform a finite element analysis of the flow in a thermocompressor.
- Purchase and/or build and bench test a thermocompressor that is optimized for absorption chillers.

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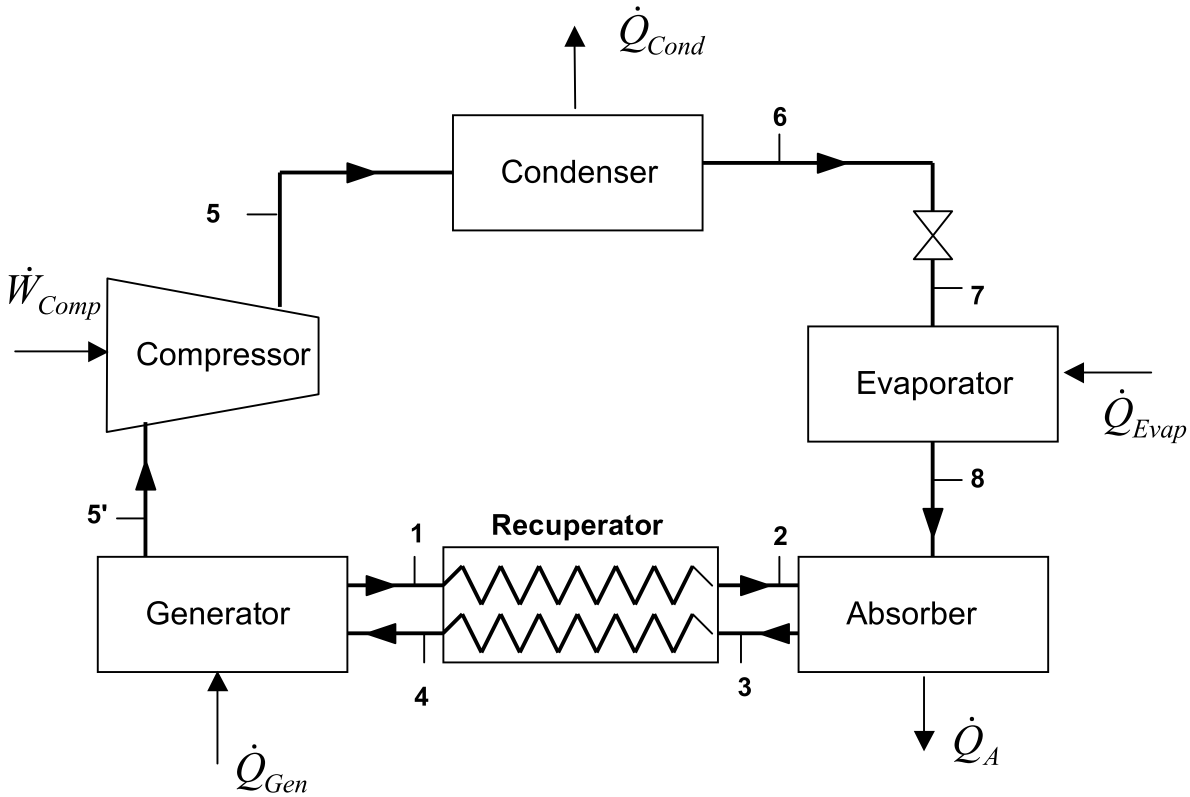
Appendix I

Single Effect Absorption Chiller Model

Single Effect State Point Description		
Point	Description	State
1	Solution Leaving Gen.	Strong Solution
2	Solution Entering Absorber	Strong Solution
3	Solution Leaving Absorber	Weak Solution
4	Solution Entering Generator	Weak Solution
5'	Refrigerant Leaving Generator	Superheated Steam
5	Refrigerant Leaving Compressor	Superheated Steam
6	Refrigerant Leaving Condenser	Saturated Liquid
7	Refrigerant Entering Evaporator	Saturated Vapor + Liquid
8	Refrigerant Leaving Evaporator	Saturated Vapor



- **Single Effect Generator**

<u>Mass Balance</u>

$$\dot{M}_{W,4} = \dot{M}_{S,1} + \dot{M}_{r,5}$$

<u>Salt Balance</u>

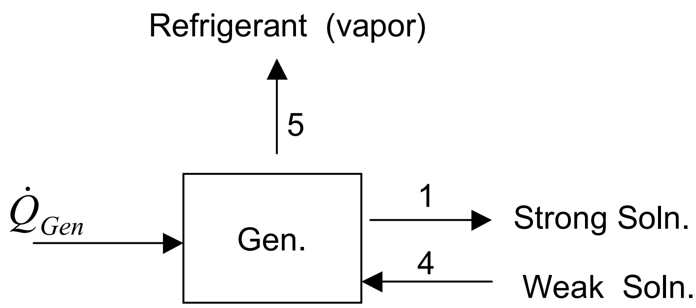


Figure 1 Generator

$$\dot{M}_{W,4}X_{W,4} = \dot{M}_{S,1}X_{S,1}$$

<u>Energy Balance</u>

$$\dot{Q}_{Gen} + \dot{M}_{W,4}h_4 = \dot{M}_{r,5}h_5 + \dot{M}_{S,1}h_1$$

By Combining the Salt and Mass Balance Equations the solution flow can be determined.

$$\dot{M}_w = \frac{\dot{M}_r}{1 - \frac{X_W}{X_S}}$$

and

$$\dot{M}_S = \dot{M}_W - \dot{M}_r$$

- **Condenser**

Mass Balance

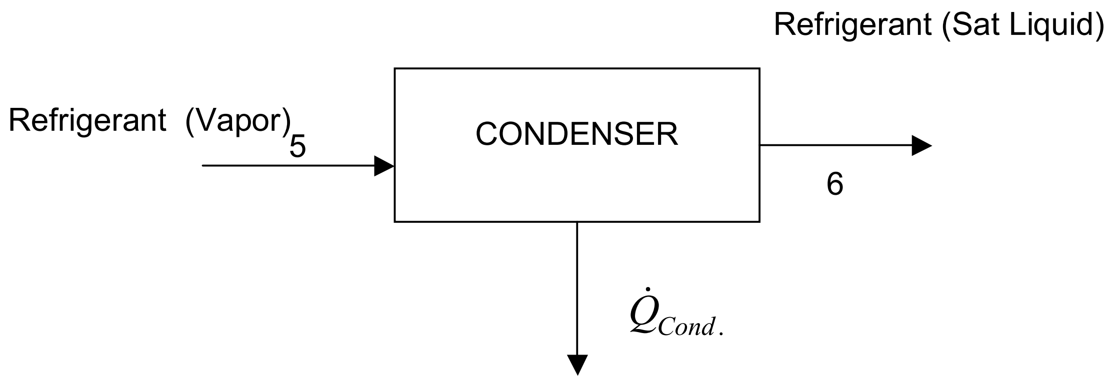


Figure 2 - Condenser

$$\dot{M}_{r,5} = \dot{M}_{r,6}$$

Energy Balance

$$\dot{M}_{r,5} h_5 = \dot{M}_{r,6} h_6 + \dot{Q}_{Cond}$$

- **Absorber**

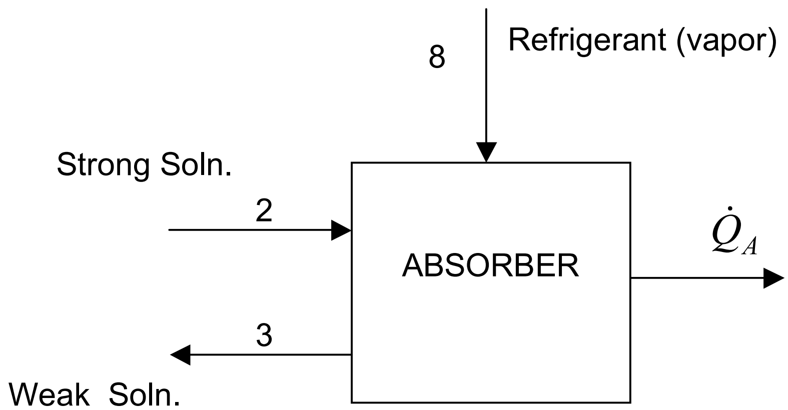


Figure 3 - Absorber

Mass Balance

$$\dot{M}_{r,8} + \dot{M}_{S,2} = \dot{M}_{W,3}$$

Salt Balance

$$\dot{M}_{S,2}X_{S,2} = \dot{M}_{W,3}X_{W,3}$$

Energy Balance

$$\dot{M}_{S,2}h_2 + \dot{M}_{r,8}h_8 = \dot{Q}_A + \dot{M}_{W,3}h_3$$

- Evaporator

<u>Mass Balance</u>

$$\dot{M}_{r,7} = \dot{M}_{r,8}$$

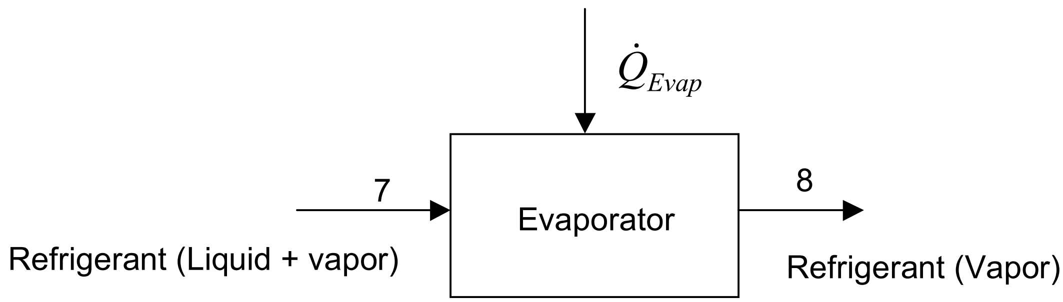


Figure 4 - Evaporator

<u>Energy Balance</u>

$$\dot{M}_{r,8} h_8 = \dot{M}_{r,7} h_7 + \dot{Q}_{Evap}$$

- Recuperator

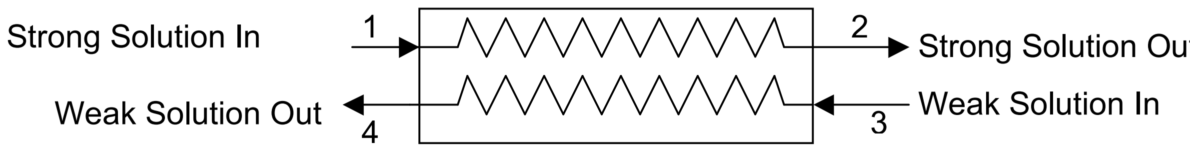


Figure 5 Recuperator

Energy Balance

$$\dot{M}_{S,1}h_1 + \dot{M}_{W,3}h_3 = \dot{M}_{S,2}h_2 + \dot{M}_{W,4}h_4$$

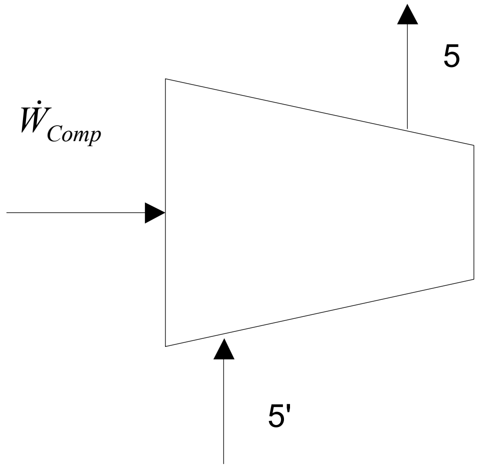
Recuperator Effectiveness

$$T_4 = T_3 + \varepsilon \frac{C_s}{C_w}\left(T_1 - T_3\right)$$

$$C_S\left(T_1 - T_2\right) = C_W\left(T_4 - T_3\right)$$

where C is the heat capacity rate defined as the product of the mass flowrate and the specific heat of the fluid. For the strong solution $C_S = \left(\dot{M}C_p\right)_S$ and for the weak solution $C_W = \left(\dot{M}C_p\right)_W$. In this application the strong solution will always have the smaller heat capacity rate.

- Compressor



Mass Balance

$$\dot{M}_{r,5} = \dot{M}_{r,5'}$$

Energy Balance

$$\dot{W}_{Comp} = \dot{M}_{r,5}\left(h_{5'} - h_5\right)$$

$$Compression\ Ratio = \frac{P_5}{P_{5'}}$$

$$\eta_{Comp} = \frac{h_{5S} - h_{5'}}{h_5 - h_{5'}}$$

Appendix II

Double Effect Absorption Chiller Model

This appendix describes a model that calculates the performance of a double effect absorption chiller. The model uses the first law of thermodynamics and the principle of conservation of mass to calculate the temperatures, pressures, flow rates, energy transfers, and coefficient of performance (COP) of the absorption chiller cycle. The chiller uses LiBr/H_20 as the absorbent and H_2O as the refrigerant. The model will be used to evaluate the effect of including a compressor with the cycle to lower the operating temperature of the high temperature generator.

Figure 1 shows all of the components of the chiller and labels the state points. Table 1 describes the state points in the double effect absorption cycle.

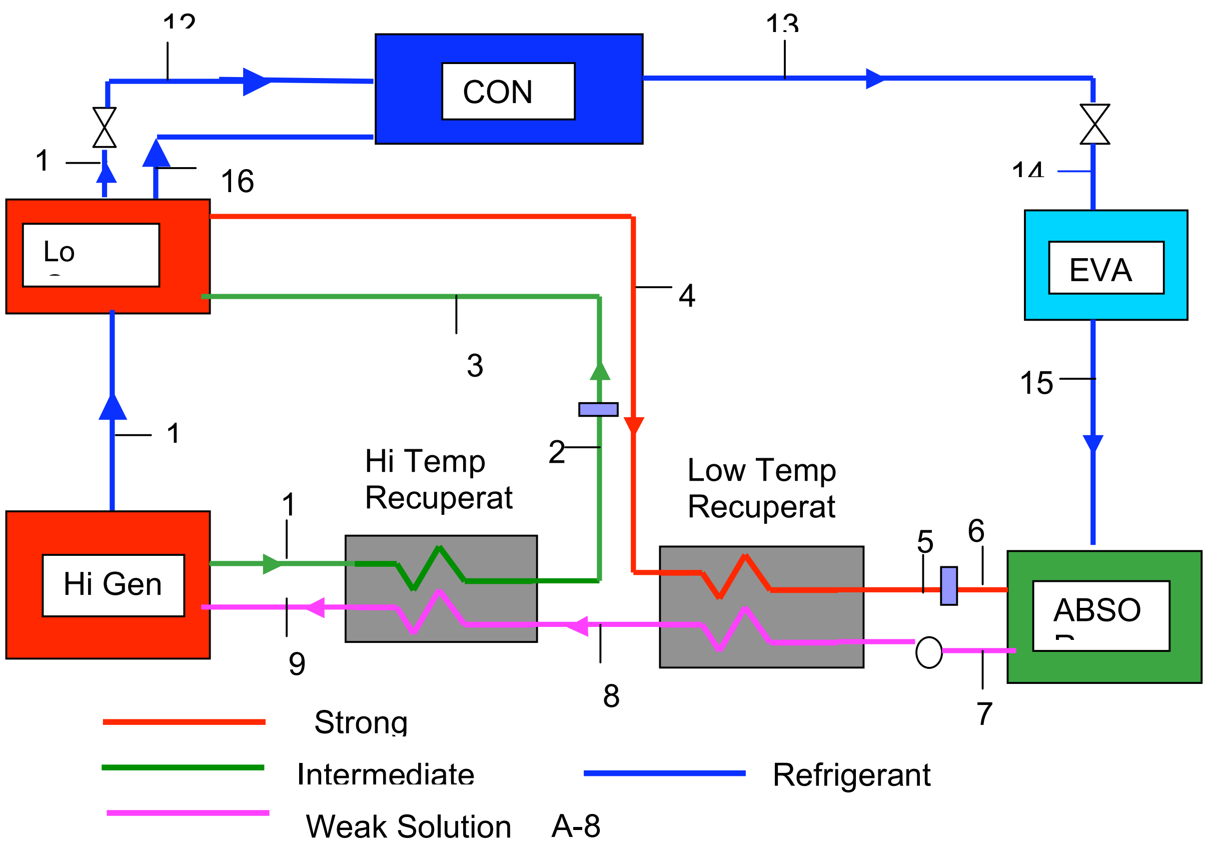


Table 1 - State Point Description

State Point	**Description**	**Fluid**
1	LiBr leaving the Hi Temp Gen.	Intermediate Solution
2	LiBr leaving Heat Exchanger #1.	Intermediate Solution
3	LiBr entering Lo Temp Gen. to create more refrigerant.	Intermediate Solution
4	LiBr leaving Lo Temp Gen.	Strong Solution
5	LiBr leaving Heat Exchanger #2.	Strong Solution
6	LiBr entering absorber to recombine with refrigerant.	Strong Solution
7	LiBr leaving the absorber.	Weak Solution
8	LiBr leaving Heat Exchanger #2.	Weak Solution
9	LiBr leaving H.X. #1. To Hi Temp Gen to start process again	Weak Solution
10	Refrigerant generated by the Hi Temp Gen to become the heat source for the Lo Temp Gen.	Superheated Steam
11	Refrigerant from high temp Gen now leaving the Lo Temp Gen after giving up it's heat to generate low temperature refrigerant	Liquid
12	Refrigerant entering the condenser.	Steam with Quality
13	Refrigerant leaving the condenser	Liquid
14	Refrigerant entering the evaporator after passing through an expansion valve	Steam with Quality
15	Refrigerant leaving the evaporator after collecting heat from the chilled water loop	Saturated Vapor
16	Refrigerant generated by the Lo Temp Gen	Superheated Steam

The equations for each of the 7 components are discussed below. Since the equations cannot be solved directly an iterative method is used. The strong and weak solution concentrations are selected by the operating temperatures of the cycle and the intermediate concentration is varied until the heat transfer within the low temperature generator balances.

- High Temperature Generator

Mass Balance

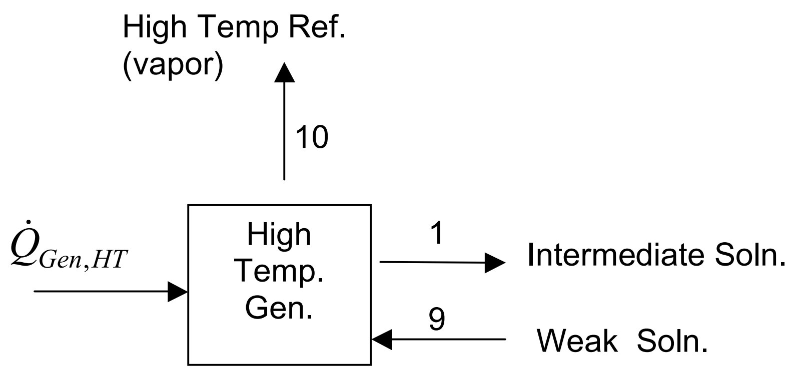


Figure 2 High Temperature Generator

$$\dot{M}_{W,9} = \dot{M}_{Htr,10} + \dot{M}_{I,1}$$

Salt Balance

$$\dot{M}_{W,9} X_{W,9} = \dot{M}_{I,1} X_{I,1}$$

Energy Balance

$$\dot{Q}_{Gen} + \dot{M}_{W,9} h_9 = \dot{M}_{Htr,10} h_{10} + \dot{M}_{I,1} h_1$$

- **Low Temperature Generator**

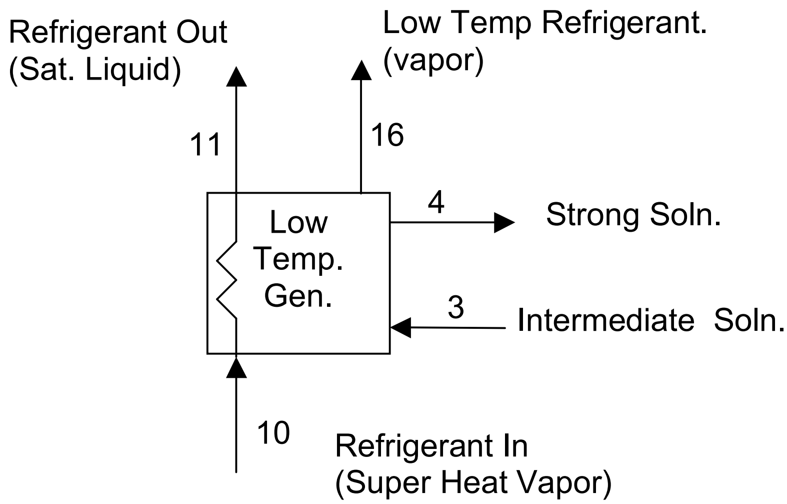


Figure 3 - Low Temperature Generator

<u>Mass Balance</u>

$$\dot{M}_{I,3} = \dot{M}_{S,4} + \dot{M}_{r,16}$$

$$\dot{M}_{r,10} = \dot{M}_{r,11}$$

<u>Salt Balance</u>

$$\dot{M}_{S,4} X_{S,4} = \dot{M}_{I,3} X_{I,3}$$

<u>Energy Balance</u>

$$\dot{M}_{Htr,10} h_{10} + \dot{M}_{I,3} h_3 = \dot{M}_{Htr,11} h_{11} + \dot{M}_{Ltr,16} h_{16} + \dot{M}_{S,4} h_4$$

- **Condenser**

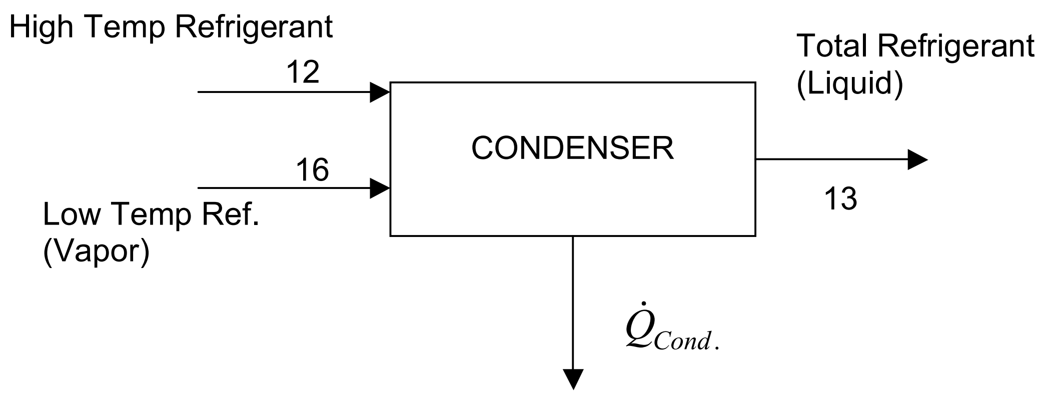


Figure 4 - Condenser

Mass Balance

$$\dot{M}_{r,12} + \dot{M}_{r,16} = \dot{M}_{r,13}$$

Energy Balance

$$\dot{M}_{Htr,12}h_{12} + \dot{M}_{Ltr,16}h_{16} = \dot{M}_{r,13}h_{13} + \dot{Q}_{Cond}$$

- Evaporator

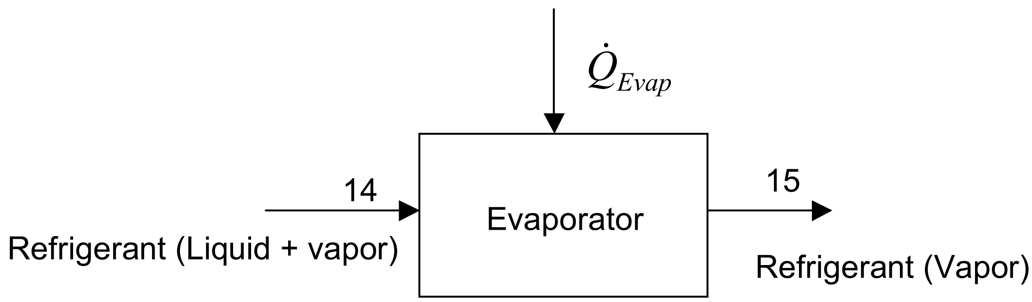


Figure 5 - Evaporator

Mass Balance

$$\dot{M}_{r,14} = \dot{M}_{r,15}$$

Energy Balance

$$\dot{M}_{r,15}h_{15} = \dot{M}_{r,14}h_{14} + \dot{Q}_{Evap}$$

- **Absorber**

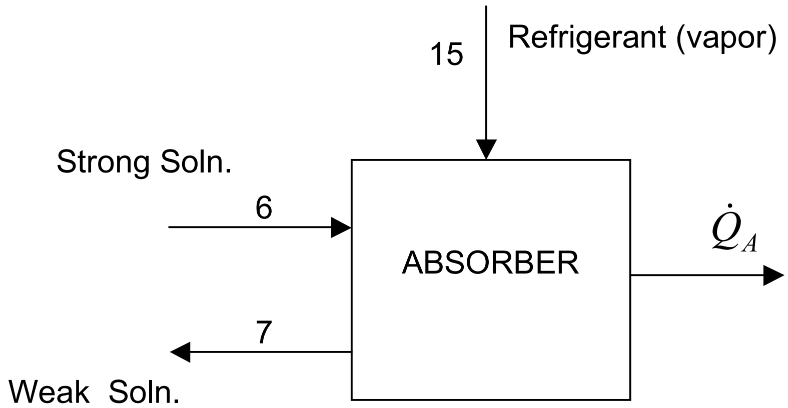


Figure 6 - Absorber

Mass Balance

$$\dot{M}_{r,15} + \dot{M}_{S,6} = \dot{M}_{W,7}$$

Salt Balance

$$\dot{M}_{S,6}X_6 = \dot{M}_{S,7}X_7$$

Energy Balance

$$\dot{M}_{S,6}h_6 + \dot{M}_{r,15}h_{15} = \dot{Q}_A + \dot{M}_{W,7}h_7$$

- High Temperature Recuperator

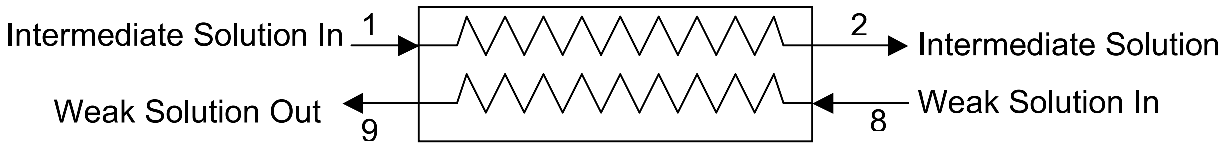


Figure 7 - High Temp Recuperator

<u>Energy Balance</u>

$$\dot{M}_{I,1}h_1 + \dot{M}_{W,8}h_8 = \dot{M}_{I,2}h_2 + \dot{M}_{W,9}h_9$$

<u>Recuperator Effectiveness</u>

$$T_2 = T_1 - \varepsilon \frac{C_I}{C_W}\left(T_1 - T_8\right)$$

The capacity rate of the intermediate solution, $C_I = \left(\dot{M}C_p\right)_I$, is always smaller than C_W.

- Low Temperature Recuperator

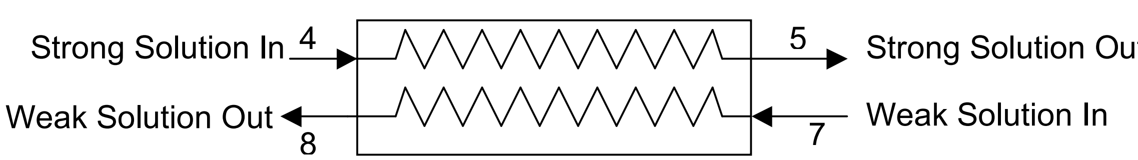
<u>Energy Balance</u>

$$\dot{M}_{S,4}h_4 + \dot{M}_{W,7}h_7 = \dot{M}_{S,5}h_5 + \dot{M}_{W,8}h_8$$

<u>Recuperator Effectiveness</u>

$$T_5 = T_4 - \varepsilon \frac{C_S}{C_W}\left(T_4 - T_7\right)$$

The strong solution capacity rate will always be smaller than the weak solution capacity rate.



There are three different locations where a compressor can be used to lower the operating temperature of a double effect absorption cycle. Figure 8 show the addition of a compressor to the absorption cycle in location 1. The compressor is placed between the high temperature generator and the low temperature generator. The compressor lowers the pressure in the high temperature generator which lowers the boiling point of the LiBr/H_2O solution within the high temperature generator.

The equations describing the compressor follow.

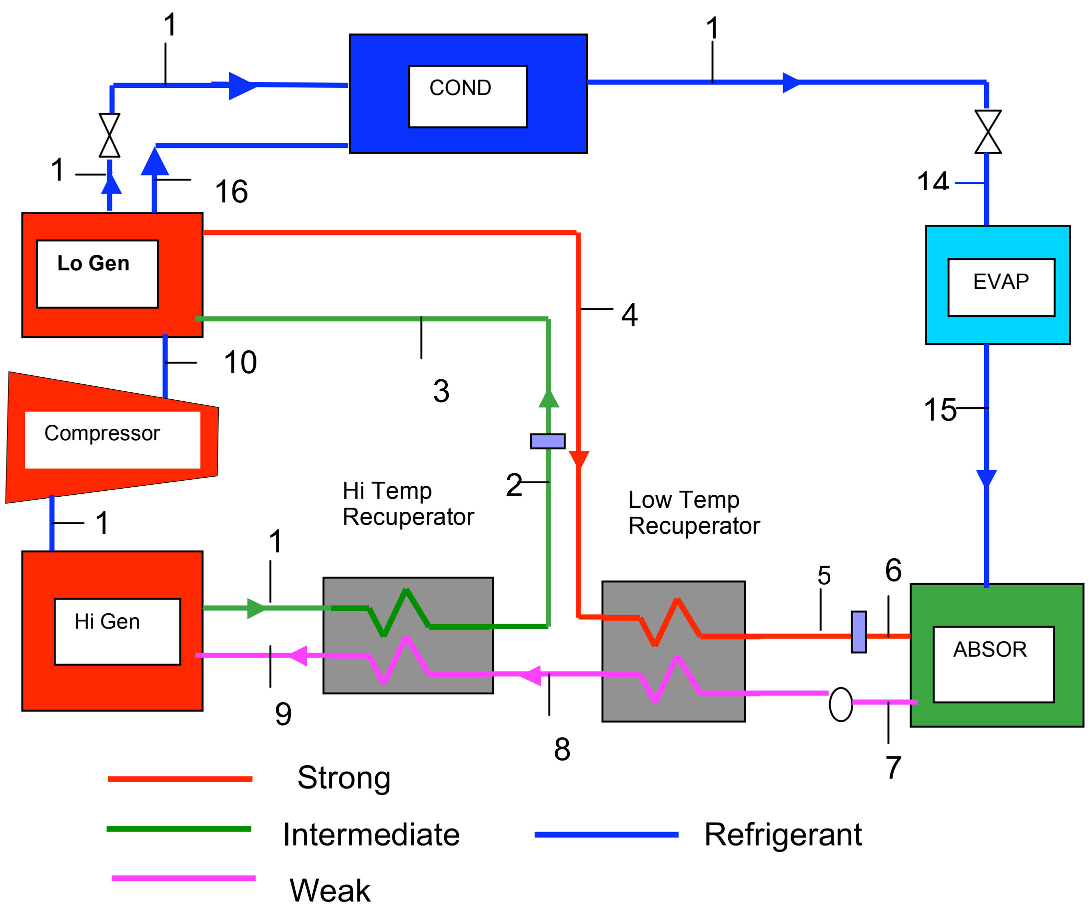
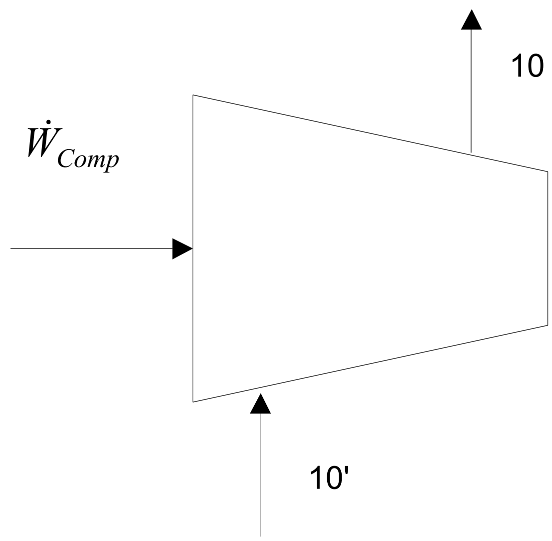


Figure 9- Double Effect Absorption Chiller

- Compressor (Location 1)



<u>Mass Balance</u>

$$\dot{M}_{Htr,10} = \dot{M}_{Htr,10'}$$

<u>Energy Balance</u>

$$\dot{W}_{Comp} = \dot{M}_{Htr}\left(h_{10'} - h_{10}\right)$$

$$Compression\ Ratio = \frac{P_{10}}{P_{10'}}$$

$$\eta_{Comp} = \frac{h_{10S} - h_{10'}}{h_{10} - h_{10'}}$$

The compressor in location 2 is shown in Figure 9.

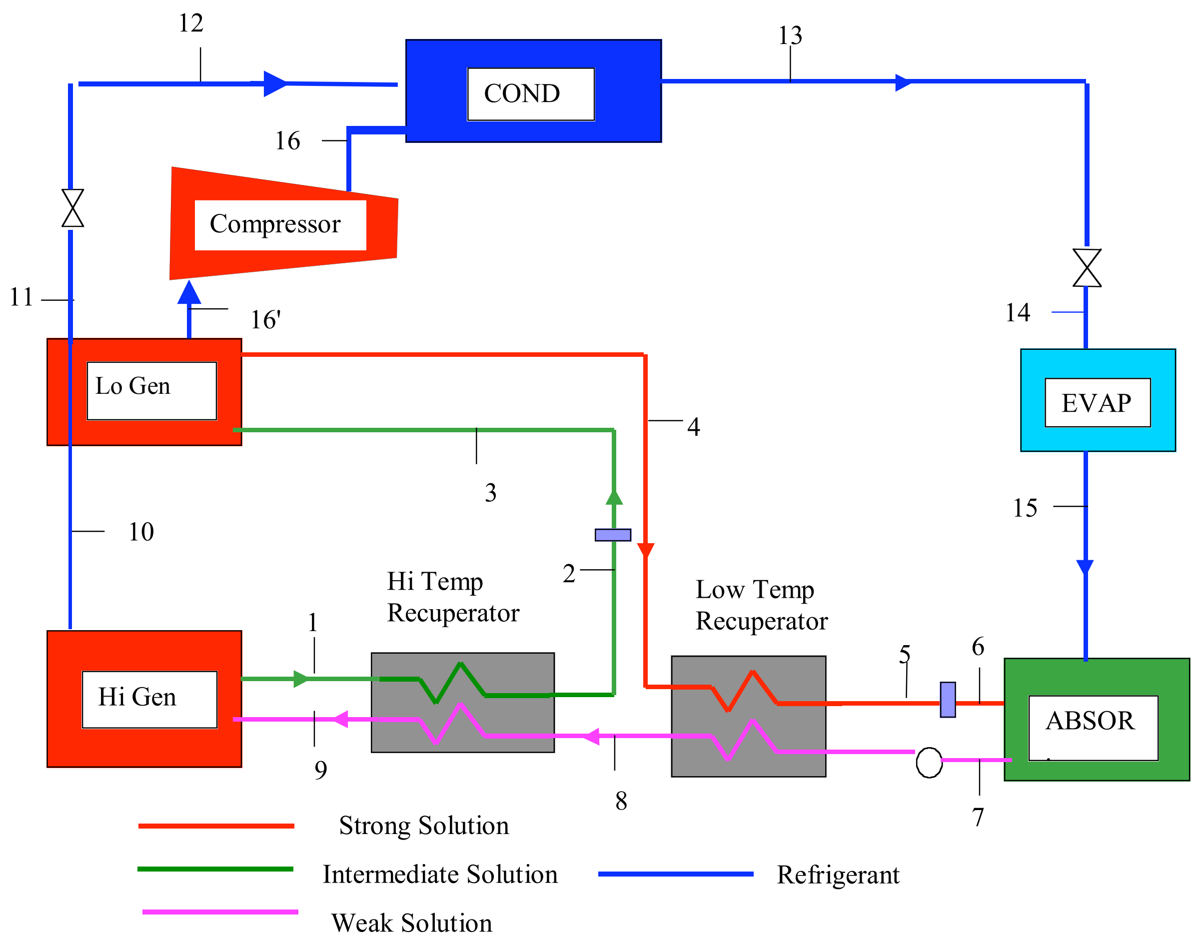


Figure 9 - Double Effect Absorption Chiller (Compressor Location 2)

Figure 10 shows the compressor in location 3.

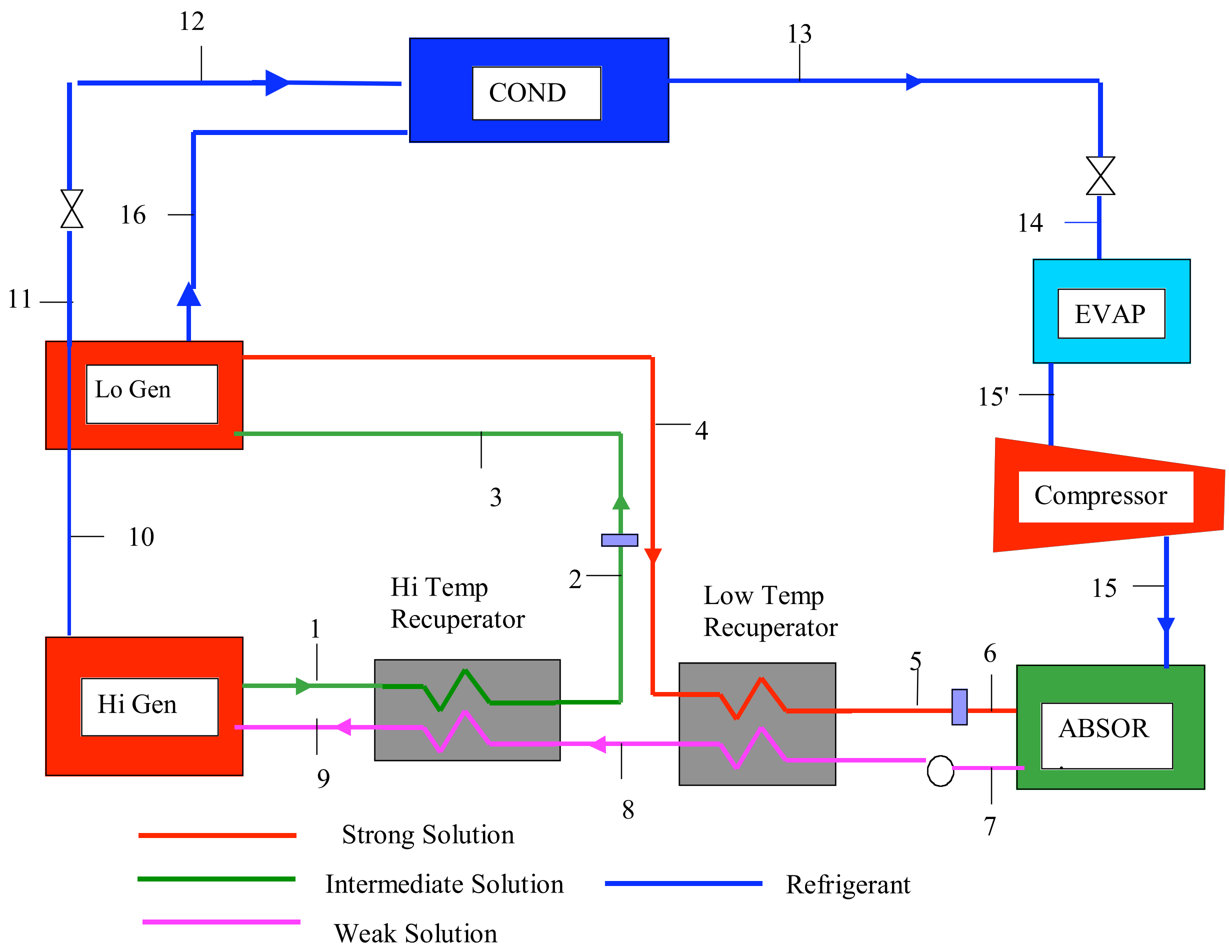


Figure 10 - Double Effect Absorption Chiller
(Compressor Location 3)